VALVE OPERATING APPARATUS AND METHOD FOR AN ENGINE

FIELD OF THE INVENTION

The present invention relates to a means and method of operating the intake and/or exhaust valves of an internal combustion engine. More particularly the invention relates to a connection arrangement between the operating apparatus and the valves, and the use of pressurised fluid to actuate valves during operation of an engine.

BACKGROUND OF THE INVENTION

Controlling the lift and timing of intake and/or exhaust (poppet) valves in an automotive engine is a necessary aspect of operating an engine. Traditionally, control has been achieved by mechanical systems using a cam to drive the stem of the intake or exhaust poppet valves, while a throttle was used to control the air flow supplied to individual cylinders. More recently, it has been known to have a cam which allows the poppet valve lift to be adjusted. However, either approach results in efficiency losses.

Many solutions have been proposed to address the efficiency loss problem. For example, mechanical solutions have been proposed to shift the phase of the camshaft or the lift of the poppet valves to improve efficiency. There have also been hydraulic-mechanical proposals attempting to shift the phase of the camshaft or lift of the valves. These systems tend to be very complicated and are not economic to manufacture.

Electro-magnetic solutions using a solenoid to drive the poppet valves have also been proposed, however a solenoid of sufficient size is relatively big, heavy and expensive, and thus not suitable for mass production.

Hydraulic solutions without a camshaft have been proposed, however the complexity of previous attempts to replace a valve train cam and throttle with a hydraulic solution has made them difficult to manufacture and thus uneconomic. The previously proposed hydraulically driven poppet valves generally include both a high and a low pressure fluid supply, requiring major modifications to an existing engine using a camshaft.

Hence, the retro-fitting of such devices is expensive because prior art proposals require the hydraulic system to be mounted in-line with the longitudinal

axis of the poppet valve and as a result, the existing cylinder head requires extensive modification. Such solutions are generally considered to only be suitable for incorporating into purpose built engines.

Some prior solutions tend to rely on a surface area difference at each end of the actuating piston, actuating the valve in one direction by the resulting force of a surface area difference. By supplying fluid constantly to a first end of the piston and alternately supplying and draining fluid to the second end, the fluid pressure at the first end, having smaller surface area, causes the piston to move when there is no pressure at the second end, and the fluid pressure multiplied by a larger surface area at the second end results in a greater force to return the piston when fluid is supplied to the second end. Accordingly, the solution involving differing surface areas remains inefficient, as the force applied by the pressure at the larger end must overcome the constant force applied by the pressure at the smaller end, slowing down the system and causing difficulty in operating at the high speeds required for a combustion engine.

For example, DE19826047A1 and WO03106820A1 disclose hydraulic actuation of the poppet valve by supplying fluid to a reciprocating piston at a nominally constant pressure. The force driving the piston, which may be approximated as Force = Pressure x Area, is dependent on the difference in surface area of each end of the piston. Actuation methods based upon this principle of operation necessarily face difficulty in reaching the high operating speeds required by an internal combustion engine.

Other prior solutions rely on constantly supplying fluid at a lower pressure to one piston end, and supplying fluid at a higher pressure to the other end to drive the piston. By alternately supplying and draining the higher pressure end of the piston, the piston is caused to reciprocate. However, this again results in slowing down the system and causing difficulty in operating at the high speeds required for a combustion engine.

Further efficiency problems exist in the above systems and in the electro-hydraulic valve operating apparatus systems disclosed in DE19826047A1, WO9207172 and US6321703. Problems experienced by such electro-hydraulic valve operating apparatus include the difficulty of sealing fast moving parts, in particular the poppet valves themselves. Where a valve stem passes through the

wall of a chamber that is required to contain fluid under pressure, a seal extending around the stem is necessarily required to prevent a significant leakage of fluid from the chamber around the valve stem. Accordingly, the stem requires a "high pressure" seal extending around the stem to prevent loss of fluid. Of course, a high pressure seal causes a significant frictional force that must be overcome in order to operate the valve, hence requiring higher fluid pressures in the valve operating apparatus in order to operate the piston. This in turn requires a stronger seal to prevent pressurised fluid from leaking out of a chamber around the poppet valve stem. The frictional force presented by a high pressure seal and the greater fluid pressure required to operate the valve results in less precision of control of the position of the poppet valve(s), less ability to provide variable control and hence lower overall engine efficiency, as the lift and timing of the poppet valves, the amounts of air being fed into the engine and hence thermodynamic efficiency of the engine cannot be controlled with the desired precision.

The prior art does not provide a simple, reliable and in particular, efficient, variable engine valve control system. Previously proposed solutions are complex and inefficient to operate. The inefficiencies of fluid pressures working against each other and the internal friction caused by high pressure seals around valve stems result in less precision of control and difficulties in achieving operation at high speed. Ultimately, the inability to operate the engine in the most efficient way inhibits the ability to operate a hydraulic valve actuating apparatus at the speeds required. The prior art is also not generally suitable for retrofitting to existing engines that include a camshaft.

As the characteristics of simplicity, reliability and efficiency are important attributes for components of complex machines such as engines, it is a primary object of the present invention to provide a means and method of operating valves that embodies one or more of these characteristics moreso than devices proposed in the prior art.

Any discussion of documents, devices, acts or knowledge in this specification is included to explain the context of the invention. It should not be taken as an admission that any of the material formed part of the prior art base or

the common general knowledge in the relevant art on or before the priority date of the claims herein.

SUMMARY OF THE INVENTION

A first aspect of the present invention provides a valve operating apparatus for an internal combustion engine including:

a housing;

a reciprocating piston residing wholly within the housing, the reciprocating piston driving one or more poppet valves;

a first fluid supply path and a first fluid drain path, each path being controllable to supply or drain fluid to/from a first reciprocating piston end;

a second fluid supply path and a second fluid drain path, each path being controllable to supply or drain fluid to/from a second reciprocating piston end;

wherein said reciprocating piston, in use, is driven between a first position and a second position by controlling said fluid in said supply and drain paths, thereby operating said one or more poppet valves, and wherein a connector passes through an aperture in said housing to connect said reciprocating piston to said one or more poppet valves, said reciprocating piston in co-operation with an internal wall of the housing forming a seal to prevent substantial egress of fluid from the housing through said aperture.

In a particularly preferred embodiment, said aperture is substantially sealed by at least a portion of the external surface of said reciprocating piston to prevent egress of fluid from the housing through said aperture.

Prior hydraulic valve operating apparatus requires a seal between the moving poppet valve stem and the hydraulic fluid supply at the point where the poppet valve stem passes through the housing. Advantageously, the present arrangement avoids such a seal. Instead, the reciprocating piston itself acts as a seal to prevent pressurised fluid from reaching the aperture from within the housing.

Internal friction in the hydraulic valve operating apparatus is lowered, as friction between the reciprocating piston and housing, already present, is not significantly increased when the reciprocating piston is used to prevent leakage of fluid through an aperture in an external wall of the housing.

In addition to reducing the number of parts required, the valve operating apparatus of the present invention operates more efficiently, with less friction as compared with prior systems. Accordingly, higher engine operating speeds may be reached as a result of the increased efficiency of operation. Furthermore, the lower internal friction means that the hydraulic system may operate at a lower fluid pressure to effect actuation of the poppet valves.

In a particularly preferred embodiment of the invention a connector rod is fixed to the reciprocating piston and connects to one or more poppet valves. The connector rod may be integrally formed with the piston, or could be a rod passing through a hole in the piston, and hence caused to move longitudinally with the piston as the piston reciprocates. The connector preferably passes through two apertures in the housing, one on each side of the piston, in order to drive two poppet valves, however an embodiment where the connector passes through a single aperture is envisaged.

In a preferred embodiment of the present invention, said first reciprocating piston end and said second reciprocating piston end have substantially the same surface area, movement of the reciprocating piston being due to the alternating supply and drainage of fluid to the piston ends. This preferred embodiment of the present invention relies on a pressure difference as the driving force in both operating directions and preferably fluid at the same nominal pressure is supplied in turn to each end of the reciprocating piston.

Although the pressures at each end of the reciprocating piston may work against each other when it is desirable to control the reciprocating piston, for example, to decelerate, in general the force (pressure) applied at one end of the reciprocating piston does not have to overcome an opposing force at the other end, that in prior systems is caused by the constant supply of fluid to one end of the piston. In prior systems, work and thus a higher pressure are required to overcome the force exerted by this fluid when the piston is to be driven against the constant pressure. Nonetheless, in a less preferred embodiment of the invention there is a difference in pressure between fluid delivered to the first reciprocating piston end and fluid delivered to the second reciprocating piston end.

Another embodiment of the present invention relies upon a constant supply of fluid being supplied to one reciprocating piston end, the piston ends being of differing surface areas so that alternating supply of fluid to the other piston end will cause the piston to reciprocate. Again, this embodiment is less preferred as overall efficiency will be lowered.

In yet another preferred embodiment of the present invention, said fluid supply and drain paths are opened and closed to control the flow of said fluid, said opening or closing of each said fluid supply and drain paths achieved by one or more solenoid valves or rotary valves, or a combination of said control valve types. These fast reacting control valves enhance the responsiveness of an apparatus according to the present invention.

In a particularly preferred embodiment of the present invention, each of said first fluid supply path, first fluid drain path, second fluid supply path and second fluid drain path has a control valve, operation of the four said control valves regulating the flow of fluid to said first and second reciprocating piston ends, thus enabling control of the movement of the reciprocating piston and hence operation of the on or more poppet valves.

Preferably, each of the four said control valves is independently operable. Preferably, each of the four said control valves may be operable to have a closed, partially open or open state.

The four control valves, one for each of the supply and drain paths, allow an extensive control of the movement of the reciprocating piston and hence the poppet valves, including the ability to operate the reciprocating piston and poppet valves at high speeds and to accelerate and decelerate the reciprocating piston to prevent valve crash or meet other valve timing objectives.

Another preferred embodiment of the invention has a reservoir of high pressure fluid in fluid connection with one or more of said fluid supply paths. Use of a reservoir may be of assistance at engine start up, in order that the engine may be started without difficulty as there is no time delay to build up pressure, as may occur when a pump alone is used. Furthermore, any momentary interruption of supply from the pressure supply device may be compensated for by the high pressure reservoir.

Controlling said fluid in said supply and drain paths may be achieved via an engine management system, also referred to as an electronic control device. Information regarding the engine speed, desired torque output, fluid and air temperatures and pressures, air humidity and inlet air mass flow and valve positions may be provided to the engine management system controller. The engine management system controller may enable variable lift and variable timing control of the one or more poppet valves.

In a preferred embodiment, the reciprocating piston may be decelerated by controlling said fluid in said supply and drain paths to avoid crashing of said one or more poppet valves onto their respective seats.

In another preferred embodiment, the reciprocating piston is biased to a predetermined position when in an inoperative state, thereby biasing each said poppet valve to a predetermined position. The biasing means may be prevented from acting on the reciprocating piston when the reciprocating piston is in an operative state.

For example, biasing means such as a spring may be provided. To ensure extra work by the hydraulic system is not required to overcome any force exerted by such a spring, the spring could optionally be snibbed in place in a compressed state while the engine is in operation, only being unsnibbed when the engine is inoperative, in order to bias the reciprocating piston and poppet valve(s) to a known position.

Further, the reciprocating piston may be partially hollow, thus providing a surface upon which vertical force may act at least at one end of said reciprocating piston.

In an above described embodiment, the connection between the reciprocating piston and the one or more poppet valves is effected by a connector rod fixed to the reciprocating piston. Preferably, the or each connection to the one or more poppet valves allows the one or more poppet valves to spin about their respective longitudinal axes.

BRIEF DESCRIPTION OF THE DRAWINGS

Preferred embodiments of the invention will now be described, without limiting the overall scope of the invention, with reference to the accompanying drawings in which:

Figure 1 shows a schematic of one embodiment of the present invention; and

Figure 1a shows a schematic of a second embodiment of the present invention; and

Figure 2 shows a cross-sectional view of a preferred embodiment of the present invention; and

Figure 3 shows a cross-sectional view rotated 90 degrees of the embodiment of Figure 2; and

Figure 4 shows an embodiment of a solenoid valve; and

Figure 5 shows yet another preferred embodiment of the present invention; and

Figure 6 shows a cross-sectional view of another embodiment of the present invention.

Figure 7 shows a cross-sectional view rotated 90 degrees of the embodiment of Figure 6.

DESCRIPTION OF PREFERRED EMBODIMENT

In a preferred embodiment of the invention, a valve operating apparatus 31 (also referred to herein as a valve train device) forms part of a hydraulic system. Alternative embodiments of the hydraulic system are shown schematically in Figures 1 and 1a, while a more detailed representation of a preferred embodiment of the invention is shown in Figure 5.

Referring to Figures 1, 1a and 5, a pressure supply device 20, for example a pump driven by the engine, delivers pressurised fluid from a sump 21 at low pressure to a high pressure reservoir 22. When the high pressure reservoir 22 is full, a pressure relief valve 23 (also referred to herein as a pressure control valve) located between the pressure supply device 20 and the high pressure reservoir 22 drains any excess fluid back to the sump 21. It should, however, be noted that a high pressure reservoir 22, while preferred, is not essential to the present invention. Furthermore, the high pressure distributor 29 schematically shown in Figure 1 may be provided as a part of the flow control valve 30 of Figure 2, or as a part of the valve operating apparatus 31 or valve train device itself. It should also be noted that a single hydraulic system could operate a number of valve operating apparatus 31 or valve train devices.

Referring now to Figures 2 and 3, or alternately to Figures 6 and 7, which show preferred embodiments of a valve operating apparatus 31 for an internal combustion engine, a simplified method of operation of the apparatus is as follows. Fluid at pressure may be supplied from either the high pressure reservoir 22 or from the pressure supply device 20, via a first fluid supply path 3, to a first chamber 11 formed between a first end 16 of the reciprocating piston 1 and housing 2. A first fluid drain path 5 connecting said first chamber 11 to a low pressure reservoir or sump 21 is closed and the pressure thereby built up in the first chamber 11 causes the reciprocating piston 1 to be hydraulically driven from a first position to a second position. At the same time, the second fluid supply path 4 is closed and the second fluid drain path 6 is open, allowing any fluid in the second chamber 10 to drain through the second drain path 6 to the sump 21. Motion of the reciprocating piston 1 assists this draining.

The reciprocating piston is then returned from the second position to the first position by a similar process, in which fluid at pressure is supplied from the high pressure reservoir 22 or from the pressure supply device 20, via a second fluid supply path 4, to a second chamber 10 formed between a second end 17 of the reciprocating piston 1 and housing 2. A first fluid drain path 6 connecting said second chamber 10 to the sump 21 is closed and the pressure thereby built up in the second chamber 10 causes the reciprocating piston 1 to be hydraulically driven from the second position to the first position. At the same time, the first fluid supply path 3 is closed and the first fluid drain path 5 is open, allowing fluid from the first chamber 11 to drain through the first fluid drain path 5 to the sump 21. Again, motion of the reciprocating piston 1 assists this draining.

Optionally (not shown), the first fluid supply path 3 and first fluid drain path 5 may pass through a single port in the housing 2. In the same way, the second fluid supply 4 and drain 6 paths may also pass through a single port.

Movement of the reciprocating piston 1 drives one or more poppet valves 7 via a connector rod 9. One or more poppet valves 7 are connected to the connector rod 9, the connection 8 being outside the housing 2.

As shown in Figures 2, 3, 5, 6 and 7 the connector rod 9 passes through an aperture 14 in the housing 2, the reciprocating piston 1 in co-operation with an internal wall of the housing 2 forming a seal to prevent substantial egress of fluid

from the housing 2 through the aperture 14. Some small amount of fluid may egress under normal operation.

In the preferred embodiment of the invention shown, the seal provided by said reciprocating piston 1 in co-operation with said internal wall of the housing 2 is located adjacent said aperture 14 such that the aperture 14 is substantially sealed to prevent egress of fluid from the housing 2 through said aperture 14. However, in an embodiment not shown, it is not necessary for the reciprocating piston 1 to abut the housing 2 at aperture 14, but simply for the reciprocating piston 1 to abut the housing 2 at locations which prevent fluid from the chambers 10 and 11 from reaching the aperture 14. Hence, a reciprocating piston 1 may not have a generally constant diameter as shown in the Figures, but may instead have a central portion of smaller diameter which forms a void annulus with the housing 2. Alternately, in a less preferred embodiment, the reciprocating piston may seal only the chamber 10 or 11 containing pressurised fluid, there being no loss of pressure caused by failing to seal an empty chamber.

The overall internal friction of the valve operating apparatus is reduced, as the friction between the reciprocating piston 1 and housing 2 will exist regardless of the location of an aperture 14 in the housing 2. In prior arrangements, the reciprocating piston drives the poppet valve by passing the poppet valve stem through an aperture at the base of the lower fluid chamber. Accordingly, a high pressure seal capable of sealing the hydraulic system at this point, under high operating speeds, is required. Achieving acceptable high speed operation is very difficult, due to the large amount of additional internal friction such a seal between the housing and poppet valve stem will create. In contrast, the present invention eliminates the need for such a seal, greatly reducing the internal friction of the system and enabling the reciprocating piston 1 to be more easily reciprocated at high speed as necessary, enhancing overall engine efficiency.

By using the reciprocating piston 1 in co-operation with an internal wall of the housing 2 to form a seal to prevent substantial egress of fluid from the housing 2 through the aperture 14, there is minimal increase in the friction already present and hence a significant overall improvement compared to prior arrangements which include a seal around the poppet valve stem. The present

system is able to move the reciprocating piston more quickly, enabling greater precision of control. This ensures that the engine operates more efficiently.

Further, in an engine including a valve operating apparatus according to the present invention, thermodynamic efficiency of the engine is improved, as a throttle, which is inherently inefficient, is not needed to control the amount of air fed to the engine. Instead, the precise control enabled due to the reduced internal friction enables variable control of the reciprocating piston 1 and hence the poppet valves 7. The variable lift and variable timing of the poppet valves 7 controls the amount of air fed. Furthermore, the reduced internal friction enables the reciprocating piston 1 to be more easily reciprocated at high speed as necessary, further enhancing overall engine efficiency and providing control over a greater range of operation.

The above described operation of the embodiment of the invention may be varied due to the precise control available in the invention. The reciprocating piston may be accelerated and decelerated by adjusting the timing of opening and closing the fluid paths, and/or by partially opening fluid paths. The ability to decelerate the reciprocating piston and thus the poppet valves ensures that crashing of the poppet valves on the valve seats may be prevented.

Furthermore, because the connection 8 to the poppet valves 7 is made externally to the housing 2, it is possible to fit the apparatus to a range of existing automotive engines with minimal changes to the cylinder heads. This minimises the re-tooling required to convert an automotive engine from a standard cam system, thereby reducing costs and making retro-fitting of the apparatus more economical and easier to perform.

The connector rod 9 of the embodiment shown in the Figures is a straight rod, however a connector of other shapes, such as a U shape or asymmetric shape may be appropriate for particular engines, possibly further reducing the changes required to be made to the cylinder heads. While the preferred embodiment has two diametrically opposed connector rods 9, the end of each passing through diametrically opposed apertures 14 in the housing 2, it is envisaged that a less preferred embodiment may have a single connector rod 9 and single aperture 14.

In the preferred embodiment, the connection 8 between the connector rod 9 and poppet valve 7 may allow each poppet valve 7 to spin naturally about their longitudinal axis. This is desirable as valve spin during engine operation acts to reduce the build up of sediments on the valve and valve seats.

The apparatus may be mounted to the side of the one or more poppet valves 7 rather than in-line, allowing multiple poppet valves 7 to be attached to a connector rod 9. Whilst potentially reducing the number of valve operating devices, a device according to the present invention also requires less height in the cylinder head than a device where the reciprocating piston 1 is in-line with the longitudinal axis of the poppet valve 7 stem.

The opening and closing of the first and second fluid supply and drain paths 3, 4, 5 and 6 is preferably performed by one or more fast acting control valves. In one preferred embodiment, each fast acting control valve is a solenoid valve such as the example depicted in Figure 4. The fast acting control valve may also be a rotary valve or combination valve system. In another preferred embodiment, the control valves 24 which open and close the respective fluid paths are a slide type.

The respective control valves may be located in the high pressure reservoir 22 or sump 21, or more preferably be respectively located in the first or second fluid supply or drain paths 3, 4, 5 and 6. By controlling the first and second fluid supply and drain paths 3, 4, 5 and 6 open, partially open or closed, movement of the reciprocating piston 1 is controlled. In turn, through the connector rod 9, movement of the one or more poppet valves 7 is controlled.

In a preferred embodiment, the control valves 24 are controlled by an electronic control device 19, which in turn is controlled by an Engine Management System (EMS). In this respect, sensors would provide information to the electronic control device 19 or the EMS, including information about the engine's speed, the driver's desired torque output (from an accelerator pedal sensor), the fluid temperatures and pressures, the valve positions, inlet air temperature, pressure and humidity and inlet air mass flow metering. Sensors to provide such information are generally used in modern Engine Management Systems. Information about the hydraulic system would also be detected, for example, the hydraulic fluid pressure would be sensed by a sensor placed in connection with

the fluid supply path(s). The electronic control device 19 or EMS would then use the information provided by sensors to adjust the poppet valve 7 lift and timing.

An apparatus according to the present invention allows for increased engine efficiency, and control of the apparatus enables the lift and timing of the poppet valves 7 to be controlled in a variable manner, responding to engine requirements. The precise control also allows the poppet valve(s) 7 to be smoothly stopped without crashing onto the respective valve seat(s).

In a preferred embodiment, a positioning spring 12 returns the reciprocating piston 1 to a predetermined known position and thus the one or more poppet valves 7 to a predetermined known position when the engine is not in use. This has the advantage of further reducing the complexity of control components required, as it is not necessary to determine the poppet valve 7 starting position when an engine is started. A benefit of having a positioning or return spring 12 is that the position at which the poppet valve 7 is held when the engine is inoperative is known and hence the position at start up is known. Thus, this approach represents a straightforward approach to determining poppet valve 7 position, further simplifying the apparatus and hence increasing its reliability, and ease of manufacture. To ensure extra work by the hydraulic system is not required to overcome such a spring 12, the spring 12 could optionally be snibbed in place in a compressed state by a catch 13 while the engine is in operation, only being unsnibbed when the engine is inoperative, in order to bias the reciprocating piston 1 and poppet valve(s) 7 to a known position.

It will be recognised by those skilled in the art that the return spring 12 could be located at various positions. For example, the return spring 12 could be located at one end of the reciprocating piston 1, residing within the housing 2, as is shown in the embodiment of Figure 5. Alternately, the return spring 12 could be mounted outside the housing 2, for example on the poppet valve 7, as is shown in the embodiment of Figure 2, or be attached to the connector rod 9 (not shown).

A benefit of having a high pressure reservoir 22 of fluid is that the engine may be started without difficulty as there is no time delay to build up pressure, as may occur when a pump alone is used. Furthermore, any momentary interruption of supply from the pressure supply device 20 may be compensated for by the high pressure reservoir 22. The reservoir 22 may form a part of the hydraulic flow

circuit as shown in Figure 1, or may not normally be a part of the flow circuit in usual operation, as shown in Figure 1a.

In a preferred embodiment, the reciprocating piston 1 may be formed with partially hollow ends 18, as shown in the embodiment of Figure 7. Holes in the walls of the hollow ends allow the entry of fluid from respective fluid supply paths into a chamber 10 or 11 formed between the reciprocating piston 1 and housing 2. Such an arrangement should substantially decrease the risk of the reciprocating piston 1 being momentarily jammed in the housing 2. However, the reciprocating piston need not be hollow, but of any geometry which presents a surface, which, in conjunction with the relevant chamber, allows the fluid to work upon the reciprocating piston in the direction of its longitudinal axis, even when the reciprocating piston is at full extension.

In a particularly preferred embodiment, the four fluid paths (first and second supply and drain paths 3, 4, 5 and 6) are individually controlled, each path having its own control valve 24 which can be controlled to be closed, partially open or open. Precise control of the reciprocating piston 1 is effected by the timing and condition of the control valves. The reciprocating piston 1 first 16 and second ends 17 have the same nominal surface area and the reciprocating piston 1 is driven by pressurised fluid alternately supplied to each piston end 16 and 17.

By opening control valve 24 located on the first fluid supply path 3, fluid is supplied to a first chamber 11 formed between a first end 16 of the reciprocating piston 1 and housing 2. The first fluid drain path 5 connecting said first chamber 11 to a low pressure reservoir or sump 21 is closed and the pressure thereby built up in the first chamber 11 causes the reciprocating piston 1 to be hydraulically driven from a first position to a second position. At the same time, the second fluid supply path 4 may be closed and the second fluid drain path 6 open, allowing any fluid in the second chamber 10 to drain through the second drain path 6 to the sump 21. Motion of the reciprocating piston 1 assists this draining.

Precise and variable control of the motion of the reciprocating piston 1 may be obtained by varying operation of the control valves 24. For example, in order to decelerate the reciprocating piston 1, and hence the poppet valves 7, the control valve 24 located on the second fluid supply path 4 may be opened or

partially opened before the reciprocating piston 1 has finished its motion. The pressure of the fluid supplied to the second chamber 10 exerts a force against the motion of the reciprocating piston 1, decelerating it. The amount to which the valve is opened and the timing is controlled by the electronic control device 19. A similar process may be followed for driving the reciprocating piston 1 in the opposite direction.

Importantly, in addition to being able to decelerate the reciprocating piston as desired, the distance travelled by the piston may be adjusted by the timing and amount of fluid supplied. For example, in order to shorten the distance travelled by the reciprocating piston 1, instead of to decelerate the reciprocating piston 1, and hence the poppet valves 7, the control valve 24 located on the second fluid supply path 4 may be fully opened before the reciprocating piston 1 has finished its motion, and the control valve 24 located on the first fluid supply path 3 may be closed before the reciprocating piston 1 has finished its motion. The pressure of the fluid supplied to the second chamber 10 exerts a force against the motion of the reciprocating piston 1, stopping it.

When supplying fluid to the first chamber 11, the degree to which the control valve 24 mounted in the fluid supply path 3 is opened will determine how quickly the chamber is filled and hence the speed at which the reciprocating piston 1 is driven.

Each of the four control valves 24 is independently operable, but is operated in coordination with the other valves, an engine management system controller 19 determining the degree to which the valve is opened and the timing of each valve in response to engine data.

The above abilities to directly control the speed, acceleration and deceleration of the reciprocating piston, as well as the distance travelled by the reciprocating piston 1, directly determine the timing and lift of the poppet valves 7, and hence the amount of air supplied to the engine. As the control may be adjusted and varied constantly, and is very responsive, the electronic control device can constantly adjust the lift and timing of the poppet valves, ensuring that an optimal air supply is provided at all times. This ensures greater overall engine efficiency.

In less preferred embodiments, a reciprocating piston 1 having different surface area on each end may be used, and, for example, pressurised fluid could be constantly supplied to one end. Furthermore, in other less preferred embodiments, fluid at a lower pressure may be constantly supplied to one piston end while pressurised fluid at a higher pressure is alternately supplied and drained from the second end. These embodiments may require a differing hydraulic system from that shown in the Figures. These embodiments still retain the advantage of reduced internal friction due to avoiding the need for a high pressure seal capable of operating at high speeds around the poppet valve stem.

As the present invention may be embodied in several forms without departing from the spirit of the essential characteristics of the invention, it should be understood that the above described embodiments are not to limit the present invention, but rather should be construed broadly within the spirit and scope of the present invention as defined in the appended claims. Various modifications and equivalent arrangements are intended to be included within the spirit and scope of the present invention.